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Comparison and analysis of engine exhaust gas energy recovery potential through various bottom cycles

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HIGHLIGHTS

- ► Direct recovery bottom cycle through secondary expansion demonstrates little, if any, positive potential for a gasoline engine.
- ► Direct recovery bottom cycle only suits to diesel engine at full load with high boost pressure.
- ► Indirect recovery bottom cycle suits to gasoline engine and diesel engine under larger range of operating conditions.
- ► Over-heated Rankine steam cycle has higher energy recovery efficiency than Brayton air cycle.
- ► Brayton air cycle recovery efficiency is limited by compression work and heat transfer quantity in heat exchanger.

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ABSTRACT

In this paper, aimed to recover engine exhaust gas energy and improve engine thermal efficiency, various means of bottom cycles for engine exhaust gas energy recovery are proposed, and those include direct recovery means through exhaust gas expansion, such as secondary expansion, and indirect recovery means through heat transfer, such as Rankine steam cycle, Brayton air cycle, etc. The performances and characteristics of each bottom cycle are studied by cycle processes calculation and then the energy recovery potentials are compared. The results show that direct recovery bottom cycle through secondary expansion demonstrates little, if any, positive potential for a gasoline engine, and it only suits to diesel engine at full load with high boost pressure. The improvement range also differs with engine speeds and energy recovery potential is low. However, indirect recovery bottom cycles have larger applied range and higher exhaust gas energy recovery potential compared to direct recovery means. In all indirect recovery bottom cycles discussed, the maximum energy recovery potentials reduce in the sequence of over-heated Rankine steam cycle, standard Rankine steam cycle, Brayton air cycle with regeneration and standard Brayton air cycle.

1. Introduction

It is well known that energy shortage and environmental pollution have become global issues of common concern. As the most widely used source of primary power for machinery critical to the transportation, construction and agricultural sectors, engine has consumed more than 60% of fossil oil. On the other hand, the amount of CO_2 gas released from engine, just for transportation applications, makes up 25% of all human activity related CO_2 emissions. Thus, energy conservation on engine is one of best ways to deal with these problems since it can improve the energy utilization efficiency of engine and reduces emissions [1,2].

Given the importance of increasing energy conversion efficiency for reducing both the fuel consumption and CO_2 gas emissions of engine, scientists and engineers have done lots of successful research aimed to improve engine thermal efficiency, including supercharge, lean mixture combustion, etc. However, in all the energy saving technologies studied, engine exhaust heat recovery (EHR) is considered to be one of the most effective means and it has become a research hotspot recently [3–5]. For example, Doyle and Patel, who are engineers of Mack Trucks Company in the United States, have designed a device for recovering exhaust gas heat based on Rankine cycle on a truck engine. The commissioning

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experiment of 450 kilometers showed that this device could save fuel consumption by 12.5% [6]. Cummins Company has also done some research on waste heat recovery on truck engines, and the results showed that engine thermal efficiency could improve by 5.4% through exhaust heat recovery [7]. James C. Conklin and James P. Szybist have designed a six-stroke internal combustion engine cycle with water injection for in-cylinder exhaust heat recovery which has the potential to significantly improve the engine efficiency and fuel economy[8]. In this paper, we have proposed several kinds of bottom cycles for engine exhaust gas energy recovery according to the characteristics of exhaust gas energy, and those include both the direct recovery means based on exhaust gas expansion, such as the novel concept of secondary expansion, and indirect recovery means based on heat transfer, such as Rankine steam cycle, Brayton air cycle and their improved means. After the theoretical analyses on the working principles of the two main kinds of bottom cycles were conducted, the bottom cycle's performances especially their exhaust gas energy recovery potentials were analyzed and compared on the basis of cycle processes calculation. All these have provided some theoretical basis for engine exhaust gas energy recovery and improving engine energy utilization efficiency.

2. Basic theory of bottom cycles for exhaust gas energy recovery

2.1. Engine energy and exergy distributions

What's the energy saving potential of engine exhaust gas energy recovery? To answer this question, the energy balance and exergy balance tests results based on a turbocharged, direct injection gasoline engine were given [9]. Fig. 1(a) illustrates the energy distribution of GDI engine under part load. As shown in the figure, energy utilization efficiency of modern gasoline engine (GDI engine) is still low. The fuel energy can be divided into several parts, that is, effective work, exhaust gas energy, coolant energy and other loss (unburned fuel energy and engine surface heat transfer). Under part load, the percentage of effective work changes from 27.8% to 33.5%, while that of exhaust gas energy varies between 23.7% and 35.8%. In most cases, exhaust gas energy almost equals effective work in quantity. However, not all the exhaust gas energy can be reused since it is a kind of low-grade energy. For this reason, engine exergy balance analysis was carried out, which is from the viewpoint of "available energy" [10,11]. As shown in the exergy distribution of GDI engine in Fig. 1(b), the percentage of exhaust gas exergy is always lower than that of effective exergy, but it is larger than the percentage of heat transfer exergy under most of conditions. Under part load, the percentage of exhaust gas exergy changes from 9.4% to 16.8%, and it demonstrates a great potential for engine fuel economy improvement by exhaust gas energy recovery.

2.2. Definition and classification of bottom cycles for exhaust gas energy recovery

According to our previous study, engine exhaust gas contains various forms of energy, and the main characteristic of exhaust gas energy is "low quality" from the viewpoint of converting into mechanical work, due to its high temperature but low pressure [12]. Generally, engine exhaust gas energy consists of thermal energy, pressure energy and kinetic energy [12,13]. Among them, pressure energy and kinetic energy belong to mechanical energy and both of them can be directly recovered by expanding. Theoretically, the recovery efficiency is limited by the friction loss of expansion device. However, the thermal energy of exhaust gas is a kind of low-grade energy and the recovery process depends on some indirect methods, such as heat transfer and thermodynamic cycle, thus the recovery process is more complex than that of direct recovery means, and the recovery efficiency is limited by cycle efficiency and heat transfer efficiency, etc. Besides, the main form of exhaust gas energy is thermal energy, which takes the largest proportion all along.

In accordance with the characteristics of exhaust gas energy, we propose various kinds of bottom cycles for exhaust gas energy recovery. The method that is based on exhaust gas direct expansion is defined as direct recovery means, such as secondary expansion, and the method based on heat transfer and thermodynamic cycle is defined as indirect recovery method, such as Rankine steam cycle, Brayton air cycle and their modified means. In other words, direct recovery method aims to recover the exhaust gas pressure energy, while the indirect recovery method aims to recover the exhaust gas thermal energy.

2.3. Working principles of bottom cycles

2.3.1. Direct recovery based on exhaust gas expansion

Limited by the expansion ratio and operating speed range of engine, usually, the exhaust gas does not fully expand in the firing cylinder. Due to the above reason, additional expansion devices, such as power turbine, additional expansion chamber, etc., have a potential to let exhaust gas further expand inside for energy recovery. As the additional expansion devices are directly connected to engine exhaust system, the level of working pressure in bottom cycle will affect the engine's pumping losses. As a result,

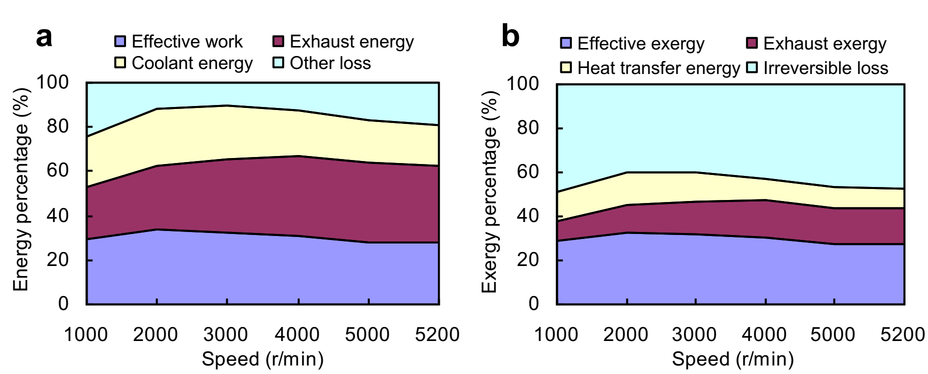

Fig. 1. Energy and exergy distributions of a turbocharged, direct injection gasoline engine. (a) Energy distribution of GDI engine. (b) Exergy distribution of GDI engine.

those bottom cycles based on exhaust gas expansion will influence the working performance of engine firing cylinder. Thus, how to match the bottom cycles with the working cycles of engine firing cylinder becomes a big technique difficulty.

2.3.2. Indirect recovery based on thermodynamic cycle

Differing from the direct recovery method of exhaust gas expansion, indirect recovery method decouples the working pressure of bottom cycle with engine exhaust back pressure by the method of heat transfer, and exhaust gas energy is converted into bottom cycle effective work through thermodynamic cycle. In these bottom cycles, the working medium plays a very important role and determines exhaust gas energy recovery efficiency. Usually, there are two main kinds of working medium forms, liquid working medium and gas working medium. On the one hand, the liquid working medium can achieve high working pressure with little compression work consumption, while the phase change process can increase the volume of working medium and in this way acquire high pressure and high volume flow rate working medium. On the other hand, when the air is used as working medium, the bottom cycle system will be much simpler since there is no phase change in heat transfer process. In addition, the bottom cycle can be designed as an open cycle which does not need a condenser, and it seems more convenient to realize on the automobile engine.

3. Evaluation of direct recovery bottom cycles through expansion

3.1. Working principle and characteristics of secondary expansion

As mentioned in the section of working principles of direct recovery bottom cycles, additional expansion devices have the potential to let exhaust gas further expand inside for energy recovery. Based on this idea, a novel concept called secondary expansion was proposed. Usually, the secondary expansion device can be either a specialized expander which is added to the exhaust gas system or the engine's cylinder. In those bottom cycles, engine exhaust gas expands again in the expander and then outputs expansion work. Fig. 2(a) and (b) illustrates the conceptual sketch of secondary expansion bottom cycle takes engine cylinders as secondary expanders. Fig. 2(a) is the schematic diagram of a two-cylinder engine without expander, and it is used for the study baseline. While in Fig. 2(b), two firing cylinders share two expanders. The firing cylinders are cylinder 1 and cylinder 4, and the expander cylinders are cylinder 2 and cylinder 3. Expander cylinder 2 and cylinder 3 have no intake valve and always open to the exhaust manifold of firing cylinder 1 and cylinder 4. The port flow coefficient is kept the same as the baseline engine, but is changed accordingly based on whether the port is used as intake or exhaust port of the expander.

Next, the working principle of this secondary expansion concept was introduced. When the exhaust valve of cylinder 1 opens, the exhaust gas will flow into cylinder 2 and cylinder 3 and then expand in the two cylinders. At the same time, cylinder 4 operates the compression stroke (both the intake valve and exhaust valve of cylinder 4 are closed). After the exhaust valve of cylinder 1 closes, the cylinder 2 and cylinder 3 operate the exhaust stroke while the cylinder 4 experiences the combustion and expansion stroke. Next, the exhaust valve of cylinder 4 opens, and at that time, its exhaust gas flows into cylinder 2 and cylinder 3 and then expands in the two cylinders. The working orders of the four cylinders are given in Table 1. As can be seen from what's mentioned above, the cylinder 1 and cylinder 4 work as four stocks, while the cylinder 2 and cylinder 3 work as two stocks. That is, cylinder 2 and cylinder 3 only experience expansion stroke and exhaust stroke. In fact, the expansion stroke is also the intake stroke. Also, the advantages of this bottom cycle can be concluded: (1) as the expander cylinder works as two strokes, it can output expansion work in each revolution; (2) this bottom cycle can be realized on automobile engine easily since it takes the engine cylinder as expander cylinder; (3) as two cylinders are used for expander cylinder, it can let the exhaust gas expand more fully. However, the intake valves of the two expander cylinders should be removed and the exhaust valves timing should also be modified in order to match the work process of expander cylinders and acquire the optimum performance.

3.2. Boundary conditions and calculation method

The boundary conditions for calculating secondary expansion bottom cycle are listed in Table 2. The base engine's displacement is 1.8 liter (4 cylinder), and its basic parameters are given in Table 3. The speeds of 1250 r/min, 2000 r/min, 4000 r/min and 6000 r/min were studied under the part load and full load, as well

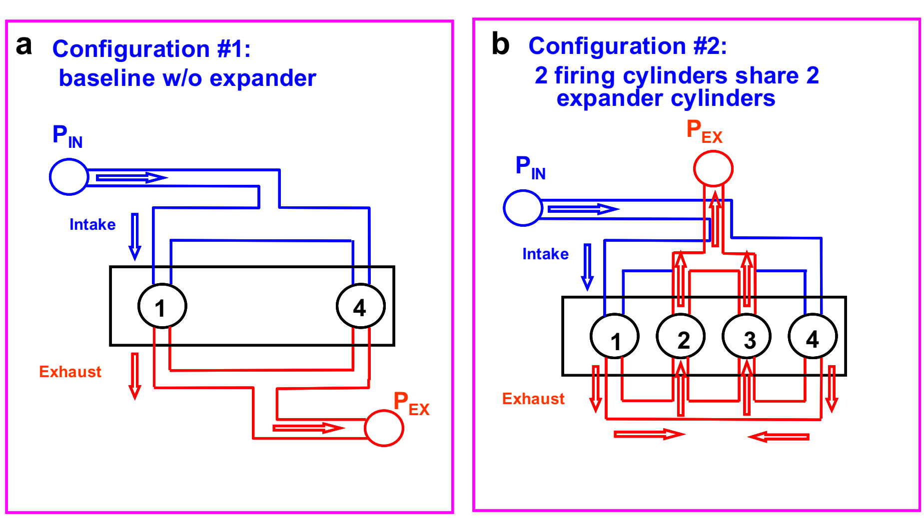

Fig. 2. Sketch of direct recovery bottom cycle: secondary expansion.

Table 1
The cylinder working orders of secondary expansion bottom cycle.

	Stroke 1	Stroke 2	Stroke 3	Stroke 4
Cylinder 1	Exhaust	Intake	Compression	Expansion
Cylinder 2	Expansion	Exhaust	Expansion	Exhaust
Cylinder 3	Expansion	Exhaust	Expansion	Exhaust
Cylinder 4	Compression	Expansion	Exhaust	Intake

as different boosting pressures. The friction mean effective pressure (FMEP) of piston type expander was acquired through expander cylinder bench test, which was conducted at the above four speeds. The result was given in Fig. 3 and it was used to estimate the FMEP in this secondary expansion bottom cycle. According to the engine basic parameters and performance data, the GT-Power model for calculating the secondary expansion concept was built, as shown in Fig. 4. In the GT-Power model, the boosting pressure was provided by the compressor, which was driven by the micromotor and/or the engine crankshaft and it depends on the boosting pressure level. In addition, the valve lift curve and timing make much difference to this secondary expansion bottom cycle. In the baseline engine, actual valve lift curve was used. When conducting different valve opening duration (event length) for the expander cylinders, valve lift was stretched (shrunk) according to the same acceleration rate as the original valve lift curve, as shown in Fig. 5. Also, during the calculation process, the exhaust valve opening timing of firing cylinder and expander cylinder were optimized. Therefore, the simulated results were the optimized value which can represent the energy saving potential of this bottom cycle scheme. Finally, after the working processes were simulated, the performances parameters of the two configurations were compared under total IMEP (firing cylinder + expander cylinder) and overall ISFC (total fuel mass flow rate divided by total indicated power of the engine).

3.3. Results and analysis

The calculation results of the baseline engine and the engine with expander under part load were given in Table 4. As can be seen from the table, for a normally aspirated gasoline engine, the secondary expansion of exhaust gas demonstrates a negative energy saving potential under part load, the additional expansion work recovered, if positive at all, cannot overcome the FMEP of the secondary expanders under nearly all the operating conditions. The lower the load (IMEP) is, the larger the negative energy saving potential will be. Only at the speed of 2000 r/min and IMEP of 12.2 bar, will the BSFC of both the baseline engine and the engine with expander come to a balance. This is because the exhaust gas pressure energy of engine is not high enough under the part load due to the intake throttling. As a result, the concept of secondary expansion bottom cycle doesn't suit to the gasoline engine part load.

Table 3
Engine parameters and calculation statement.

	Baseline W/O expanders	Configuration #2: 2 firing cylinders sharing 2 expander cylinders	
	Firing cylinders 1 and 4	Firing cylinders 1 and 4	Expander cylinders 2 and 3
Bore (mm)	82	82	82
Stroke (mm)	85	85	85
ε	10.5	10.5	10.5
Intake valve	Base engine intake	Base engine intake	No valve
Exhaust valve	Base engine exhaust, duration studied	Base engine exhaust	Base engine intake, duration studied
Intake port	Base engine intake	Base engine intake	Base engine exhaust
Exhaust port	Base engine exhaust	Base engine exhaust	Base engine intake

Table 5 gives the calculation results of the baseline engine and the engine with expander under full load and various boosting pressures. As shown in the table, the secondary expansion of exhaust gas demonstrates a negative energy saving potential even under full load (intake pressure = 1.0 bar abs.). It means that the additional expansion work recovered also can't overcome the FMEP of the secondary expanders, while the loss (negative energy saving potential) under full load is much less than that under part load. However, the secondary expansion of exhaust gas demonstrates an energy saving potential only for boosted engine under full load conditions (intake pressure >1.0 bar abs.). The higher the boosted level is, the greater the energy saving potential will be. This is because the higher boosting pressure results in higher exhaust gas pressure. Under the circumstances, exhaust gas contains more pressure energy which is output by secondary expansion in the expander cylinder. At the engine speed of 2000 r/min and boosting pressure of 2.0 bar, the BSFC improvement over two-cylinder baseline engine can reach 5.33%. All that shows a considerable energy saving potential compared to other advanced techniques on engine. As the vehicle engine especially passenger car engine seldom or never operates under these conditions, this bottom cycle concept is not suitable for passenger car engine. However, this concept fits well for heavy diesel engine running under full load at all times especially in stationary applications, e.g. Genset.

The secondary expansion changes not only the engine pumping process (either add or reduce pumping loss depends on the gas pressure level and engine speed), but also the residual gas fraction (RGF) trapped in cylinder, therefore the amount of fresh charge induced and trapped in cylinder to burn at the exhaust valve closure point of the firing cylinders. At part load (lower engine intake pressure or partial open throttle), since the firing cylinder gas pressure is not high enough during the gas exchange process, the expander cylinders behave more like a suction pump instead of an expander, i.e. no positive expansion work (even under indicated basis) could be generated in the expanders. Increase of engine intake pressure (boosting level) or running at a higher engine speed would change the situation, where the expander starts to generate

Table 2
Boundary conditions of secondary expansion bottom cycle.

Engine speed (r/min)	PIN (bar abs.)	PEX (bar abs.)	Relative AFR λ (−)	Spark timing (−)	Pressure ratio by supercharger (−)	Engine configuration (−)
1250, 2000, 4000, 6000	0.5	1.017	1	As baseline	1	NA: Part load
	0.7	1.017	1	As baseline	1	NA: Part load
	0.8	1.017	1	As baseline	1	NA: Part load
	1.0	1.017	0.9	As baseline	1	NA: full load
	1.5–2.0	1.017	0.9	5° CA retard	1.5–2.0	SC: full load
	3.0	2.5	0.9	5° CA retard	2	SC + ECT: full load

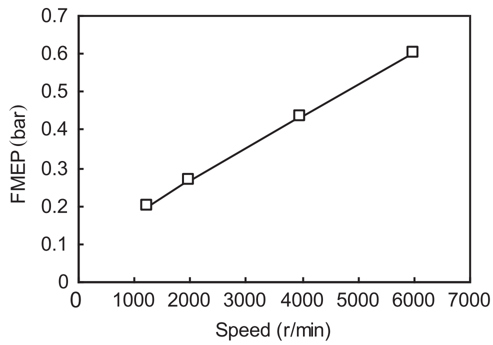

Fig. 3. FMEP of additional expander.

positive expansion work (in indicated basis). Because of all above, compared under the same intake pressure level (same throttle position or boost pressure level), although the engine with the secondary expansion would produce higher IMEP (or indicated power) due to lower trapped residual gas fraction (RGF) and therefore more fresh charger to burn, it does not necessarily improve the engine thermal efficiency.

For part load engine operation (intake pressure <1.0 bar abs., which is the most frequent operating points in an automotive application), if compared under the same engine IMEP level, the engine with secondary expansion does not show any potential to improve engine thermal efficiency, even under indicated basis (ISFC), in most of cases evaluated it shows higher ISFC due to additional pumping loss introduced by the expander. BSFC will be higher when additional FMEP is added to the expander cylinders. Only when the engine is running at full load with higher boosting pressures can the additional expander concept start to show potential to improve engine thermal efficiency, and the improvement range also differs with engine speeds. In general, there is less improvement at low speed (more important speeds for a vehicle application) but more at high speeds. However, the potential to engine brake efficiency improvement is further reduced by the fact that the additional FMEP level introduced by the expander cylinders increases with engine speed.

According to the above analyses, the following consequences are acquired: since the bottom cycle process of direct recovery is directly related to the engine exhaust gas pressure, and the exhaust gas pressure directly affects the engine exhaust process and pumping loss, the bottom cycle process and firing cylinder work process are coupled and interact with each other. Such concepts are only applicable to the high load conditions of heavy engine with high boosted pressure, and the effective work range and energy recovery potential are very limited.

4. Evaluation of indirect recovery bottom cycles through heat transfer

4.1. Indirect recovery bottom cycle: Rankine steam cycle

4.1.1. Standard Rankine steam bottom cycle

4.1.1.1. Working principle and characteristics of Rankine steam bottom cycle. As is mentioned in the working principles of bottom cycles, indirect recovery bottom cycles aim to recover exhaust gas thermal energy based on heat transfer rather than exhaust gas direct expansion. Compared to direct recovery bottom cycles, indirect recovery bottom cycles are more complex. Usually, indirect recovery methods depend on thermodynamic cycle, such as Rankine cycle, Brayton cycle, etc.

Rankine cycle is widely used in engine exhaust gas energy recovery [14–17]. In this paper, standard Rankine steam bottom cycle has been discussed firstly. It is well known that working medium is very important to the cycle efficiency and performance. The ideal working medium should meet some requirements, such as low price, non-toxic, non-polluting, stable physical and chemical

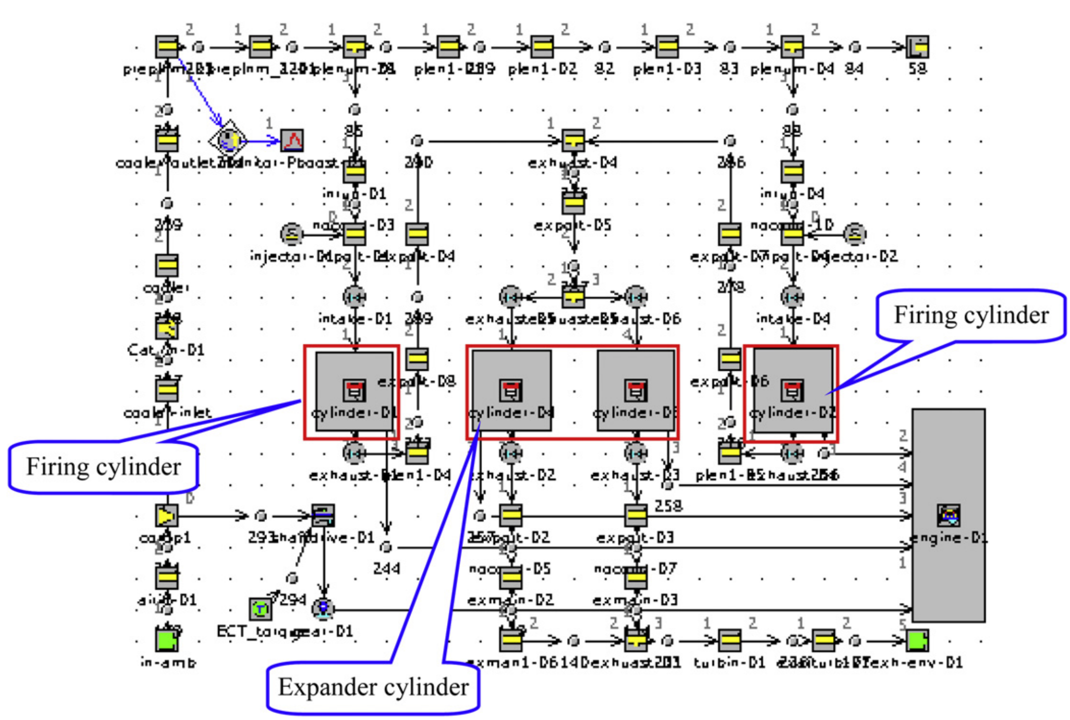

Fig. 4. GT-Power model for the secondary expansion.

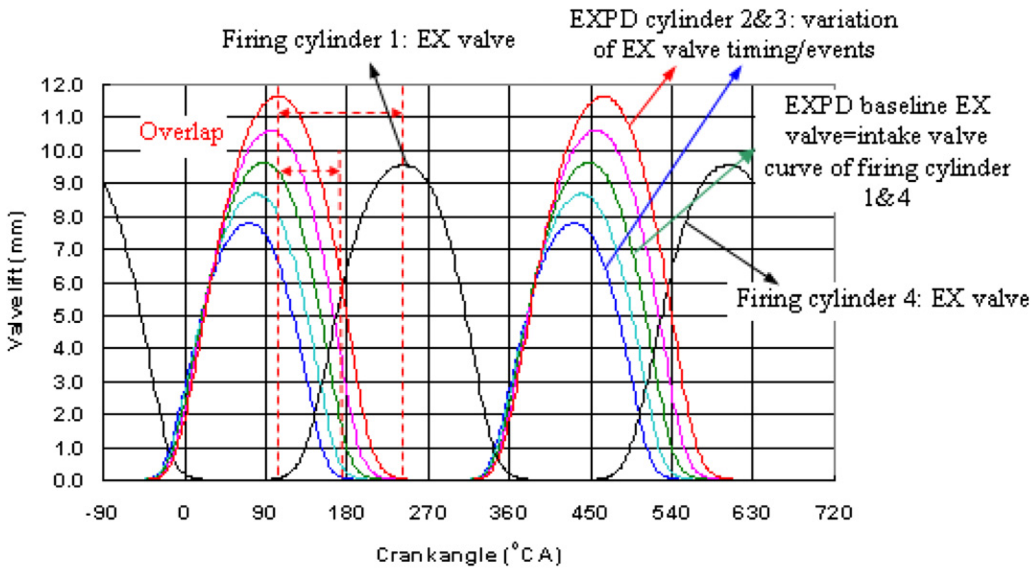

Fig. 5. Variation of expander cylinder exhaust valve lifts/event lengths.

properties at high temperature, etc. In this study, water has been used for the Rankine cycle working medium since it can meet the above requirements. What's more, water is the most common working medium of Rankine cycle in industry and the application technology is quite mature.

As shown below, Fig. 6(a) is the conceptual sketch of standard Rankine steam bottom cycle for exhaust gas energy recovery and Fig. 6(b) shows its T–S diagram of cycle processes. This bottom cycle consists of four core components, that is, pump, boiler, turbine and condenser. The working principle of Rankine steam bottom cycle is described as follows. Because the temperature of engine exhaust gas is very high, exhaust gas was used for the heat source of this thermodynamic cycle system. In the bottom cycle, firstly, the working medium water is boosted to a high pressure in the pump; then, the liquid working medium is heated by the engine exhaust gas in the heat exchanger and turns into saturated steam; next, the saturated steam expands in the turbine and outputs effective work; finally, the exhaust steam of turbine is condensed into liquid saturated water and then flows into the pump to perform the next cycle. The purpose of recovering exhaust gas energy could be realized through the four processes above. All the thermodynamic processes were illustrated in the T–S diagram of Fig. 6(b).

The main characteristics of this bottom cycle system can be listed as follows. (1) As the compressibility of liquid working medium (water) is very low, it only needs little compression work to obtain a required pressure. (2) For the main form of exhaust gas energy is thermal energy rather than pressure energy, it will have

Table 4
Potential evaluation of secondary expansion: part load.

Operating condition	2 cylinder baseline								2 cylinder + 2 expander engine								
Speed (r/min)	IMEP (bar)	ISFC (g/kWh)	FMEP (F-cyl) (bar)	FMEP (expander) (bar)	Total FMEP (bar)	BMEP (bar)	Power kW	BSFC (g/kWh)	IMEP_total (bar)	ISFC (g/kWh)	FMEP (bar)	FMEP (expander) (bar)	Total FMEP (bar)	BMEP (bar)	Power kW	BSFC (g/kWh)	BSFC_improvement over 2 cylinder baseline (%)
1250	3.0	290.0	1.057	0.00	1.06	1.94	1.8	447.8	3.0	318.0	1.057	0.20	1.26	1.74	1.6	547.3	−18.19
	4.0	272.5	1.057	0.00	1.06	2.94	2.8	370.4	4.0	299.0	1.057	0.20	1.26	2.74	2.6	436.0	−15.06
	5.0	261.0	1.057	0.00	1.06	3.94	3.7	331.0	5.0	284.0	1.057	0.20	1.26	3.74	3.5	379.4	−12.76
	7.0	251.0	1.057	0.00	1.06	5.94	5.6	295.6	7.0	269.5	1.057	0.20	1.26	5.74	5.4	328.5	−10.00
	10.6	269.0	1.057	0.00	1.06	9.49	8.9	299.0	11.1	271.0	1.057	0.20	1.26	9.83	9.2	305.6	−2.19
2000	4.0	269.0	1.191	0.00	1.19	2.81	4.2	383.1	4.0	287.0	1.191	0.26	1.45	2.55	3.8	450.9	−15.05
	5.0	258.0	1.191	0.00	1.19	3.81	5.7	338.7	5.0	272.0	1.191	0.26	1.45	3.55	5.3	383.5	−11.70
	7.0	245.5	1.191	0.00	1.19	5.81	8.7	295.8	7.0	252.0	1.191	0.26	1.45	5.55	8.3	318.1	−6.99
	8.0	242.0	1.191	0.00	1.19	6.81	10.2	284.3	8.0	248.0	1.191	0.26	1.45	6.55	9.8	303.1	−6.19
	12.2	260.0	1.191	0.00	1.19	10.96	16.4	288.3	11.9	253.0	1.191	0.26	1.45	10.45	15.6	288.2	0.03
4000	3.0	279.0	1.585	0.00	1.59	1.42	4.3	591.5	3.0	290.0	1.585	0.43	2.02	0.98	2.9	884.7	−33.14
	4.0	262.5	1.585	0.00	1.59	2.42	7.3	434.8	4.0	272.0	1.585	0.43	2.02	1.98	5.9	548.5	−20.74
	5.0	252.5	1.585	0.00	1.59	3.42	10.2	369.7	5.0	257.5	1.585	0.43	2.02	2.98	8.9	431.6	−14.33
	7.0	240.0	1.585	0.00	1.59	5.42	16.2	310.2	7.0	240.0	1.585	0.43	2.02	4.98	14.9	337.1	−7.97
	9.6	259.0	1.585	0.00	1.59	8.02	24.0	310.2	10.0	250.0	1.585	0.43	2.02	7.93	23.7	313.5	−1.05
6000	4.0	253.5	2.181	0.00	2.18	1.82	8.2	557.4	4.0	267.0	2.181	0.60	2.78	1.22	5.5	876.1	−36.37
	5.0	242.5	2.181	0.00	2.18	2.82	12.7	430.1	5.0	251.5	2.181	0.60	2.78	2.22	10.0	566.7	−24.10
	6.0	235.0	2.181	0.00	2.18	3.82	17.2	369.2	6.0	238.0	2.181	0.60	2.78	3.22	14.5	443.6	−16.77
	7.0	230.0	2.181	0.00	2.18	4.82	21.7	334.1	7.0	230.5	2.181	0.60	2.78	4.22	18.9	382.4	−12.64
	10.3	243.0	2.181	0.00	2.18	8.12	36.5	308.3	10.6	237.5	2.181	0.60	2.78	7.77	34.9	322.5	−4.41

Table 5
Potential evaluation of secondary expansion: full load.

Operating condition	2 cylinder baseline											2 cylinder + 2 expander engine									
Speed (r/min)	P_intake (bar)	IMEP (bar)	ISFC (g/kWh)	SC PR	FMEP (cyl) (bar)	FMEP-SC (bar)	FMEP (expander) (bar)	Total FMEP (bar)	BME (bar)	Power kW	BSFC (g/kWh)	IMEP_total (bar)	ISFC (g/kWh)	FMEP (bar)	FMEP-SC (bar)	FMEP (expander) (bar)	Total FMEP (bar)	BMEP (bar)	Power kW	BSFC (g/kWh)	BSFC_improvement over 2 cylinder baseline (%)
1250	1.0	10.6	269.0	1.0	1.057	0.00	0.00	1.06	9.49	8.9	299.0	11.1	271.0	1.057	0.00	0.20	1.26	9.83	9.2	305.6	−2.19
	1.5	16.9	280.0	1.5	1.057	1.00	0.00	2.06	14.79	13.9	319.0	17.4	275.0	1.057	1.00	0.20	2.26	15.14	14.2	316.0	0.93
	3.0	33.8	272.0	2.0	1.057	3.56	0.00	4.61	29.19	27.3	315.0	34.8	263.5	1.057	3.56	0.20	4.81	29.97	28.0	305.8	3.01
2000	1.0	12.2	260.0	1.0	1.191	0.00	0.00	1.19	10.96	16.4	288.3	11.9	253.0	1.191	0.00	0.26	1.45	10.45	15.6	288.2	0.01
	2.0	24.9	263.0	2.0	1.191	2.38	0.00	3.57	21.33	31.9	307.1	25.7	248.0	1.191	2.38	0.26	3.84	21.86	32.7	291.5	5.33
	3.0	36.7	264.0	2.0	1.191	3.56	0.00	4.75	31.95	47.8	303.2	36.1	250.2	1.191	3.56	0.26	5.01	31.08	46.5	290.5	4.38
4000	1.0	9.6	259.0	1.0	1.585	0.00	0.00	1.59	8.02	24.0	310.2	10.0	250.0	1.585	0.00	0.43	2.02	7.93	23.7	313.5	−1.06
	2.0	20.1	259.0	2.0	1.585	2.38	0.00	3.97	16.13	48.3	322.7	20.8	245.0	1.585	2.38	0.43	4.40	16.35	48.9	310.9	3.79
	3.0	28.7	264.0	2.0	1.585	3.56	0.00	5.14	23.56	70.5	321.6	29.5	250.3	1.585	3.56	0.43	5.57	23.89	71.5	308.7	4.20
6000	1.0	10.3	243.0	1.0	2.181	0.00	0.00	2.18	8.12	36.5	308.3	10.6	237.5	2.181	0.00	0.60	2.78	7.77	34.9	322.5	−4.41
	2.0	22.0	239.5	2.0	2.181	2.38	0.00	4.56	17.39	78.1	302.4	22.2	229.5	2.181	2.38	0.60	5.16	17.04	76.5	299.1	1.10
	3.0	31.7	243.0	2.0	2.181	3.56	0.00	5.74	25.96	116.6	296.7	31.7	235.2	2.181	3.56	0.60	6.34	25.36	113.9	293.9	0.95

a higher recovery efficiency by heat transfer than exhaust gas directly expanding. (3) Compared with the traditional method of exhaust gas secondary expansion (directly recovering exhaust gas pressure energy), the biggest advantage of this bottom cycle system is that it can decouple the pressure and temperature by the means of heat transfer. That is to say, exhaust gas energy transfers into bottom cycle working medium under the premise of hardly bringing extra exhaust gas back pressure to engine. As a result, the bottom cycle does not directly affect the working performance of ignition cylinders.

4.1.1.2. Thermodynamic processes analysis of Rankine steam bottom cycle. Next, the thermodynamic processes of Rankine steam bottom cycle were analyzed by combining with the steam T–S diagram shown in Fig. 6(b), and then the main performance parameters as well as their calculation formulas were given.

(1) In the pump, the liquid working medium is compressed to a certain pressure, and this process corresponds to process 1–2(2′) in the T–S diagram. Process 1–2′ represents the ideal process (isentropic compression) while process 1–2 is the real compression process.

$$h_2 = h_1 + \frac{W_P}{q_{m\_s}} \tag{1}$$

$$W_P = \frac{p_2 - p_1}{\rho_1} \cdot \frac{q_{m\_s}}{\eta_P} \tag{2}$$

where h_1 and h_2 are the specific enthalpy of water at state point 1 (pump inlet) and state point 2 (pump outlet), respectively, during the cycle processes; W_P represents the compression work consumed by the pump; q_{m_s} is the mass flow rate of working medium steam; p_1 is the working medium pressure at the inlet of pump while p_2 is the working medium pressure at the outlet of pump; ρ_1 is the working medium density at the inlet of pump; η_P represents the efficiency of pump.

(2) In the heat exchanger (boiler), liquid water is heated by the exhaust gas heat and turns into saturated steam. This process corresponds to the process 2–3 which is regarded as an isobaric process in the T–S diagram. At the same time, engine exhaust gas is cooled from state point 5 to state point 6. According to the energy conservation law, the following formulas are given.

$$\dot{\Phi}_{2-3} = q_{m\_s} \cdot (h_3 - h_2) \tag{3}$$

$$\dot{\Phi}_{5-6} = q_{m\_e} \cdot (h_5 - h_6) \tag{4}$$

$$\dot{\Phi}_{2-3} = \dot{\Phi}_{5-6} \cdot \eta_h \tag{5}$$

where $\dot{\Phi}_{2-3}$ is the heat flux flows into bottom cycle working medium; $\dot{\Phi}_{5-6}$ is the heat flux flows out of engine exhaust gas; η_h is the heat transfer effectiveness; h_3 is the specific enthalpy of steam at state point 3 (heat exchanger outlet); h_5 and h_6 are the specific enthalpy of engine exhaust gas at state point 5 and state point 6, respectively; q_{m_e} is the mass flow rate of engine exhaust gas.

The efficiency of counter flow heat exchanger can be calculated according to formula (6) or formula (7).

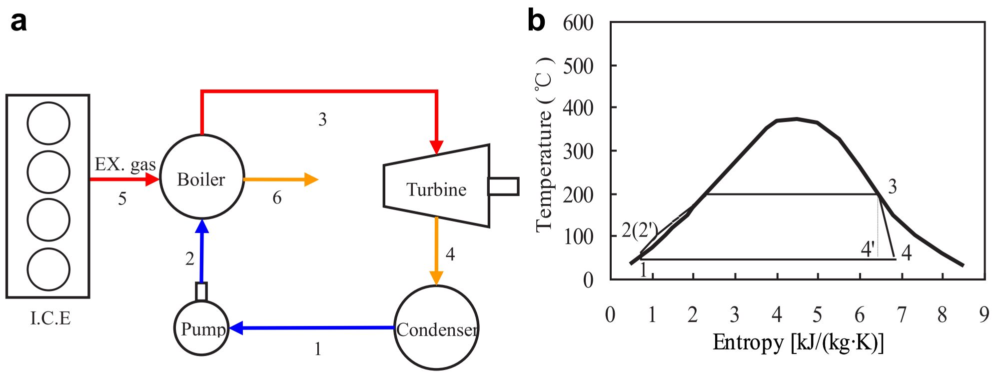

Fig. 6. Standard Rankine steam bottom cycle: working principle and T–S diagram.

$$\varepsilon = 1 - \exp(-\mathrm{NTU}) \tag{6}$$

$$\varepsilon = \frac{1 - \exp\left\{(-\mathrm{NTU})\left[1 - \frac{(q_m c)_{min}}{(q_m c)_{max}}\right]\right\}}{1 - \frac{(q_m c)_{min}}{(q_m c)_{max}}\exp\left\{(-\mathrm{NTU})\left[1 - \frac{(q_m c)_{min}}{(q_m c)_{max}}\right]\right\}} \tag{7}$$

$$\mathrm{NTU} = \frac{kA}{(q_m c)_{min}} \tag{8}$$

where ε is the efficiency of heat exchanger; NTU is the number of transfer unit; $(q_m c)_{min}$ and $(q_m c)_{max}$ are the heat capacities of the two kinds of fluids; k is the coefficient of heat transfer; A is the heat transfer area. If the phase of working medium changes during the heat transfer process in the heat exchanger, the efficiency of heat exchanger can be calculated according to formula (6); or else, formula (7) is used to calculate the efficiency of heat exchanger.

(3) In the turbine, working medium steam expands and outputs effective work, and this process corresponds to process 3–4(4′) in the T–S diagram. Process 3–4′ represents the ideal process (isentropic expansion), while process 3–4 is the real process (irreversible expansion).

$$W_{T_1} = q_{m\_s} \cdot (h_3 - h_{4'}) \cdot \eta_T = q_{m\_s} \cdot (h_3 - h_4) \tag{9}$$

where W_{T_1} represents the steam expansion work of Rankine steam cycle; η_T is the isentropic efficiency of turbine; h_4 is the specific enthalpy of steam at state point 4 (real process), while $h_{4'}$ is the specific enthalpy of steam at state point 4′ (isentropic process).

(4) In the condenser, working medium steam is condensed into liquid saturated water, and this process corresponds to process 4–1 in the T–S diagram. Usually, it can be regarded as an isobaric process.

$$W_{Co} = q_{m\_s} \cdot H_{Co} = q_{m\_s} \cdot (h_1 - h_4) \tag{10}$$

where W_{Co} represents the total heat released in the condenser, H_{Co} is the enthalpy drop of unit mass working medium.

(5) In Rankine steam bottom cycle, effective output work equals the expansion work of turbine minus the compression work of pump. By combining with the formulas (2), (3) and (9), the calculation formula for the thermal efficiency of Rankine steam cycle is given in formula (11).

$$\eta_{t\_R} = \frac{W_{net}}{\dot{\Phi}_{2-3}} \cdot 100\% = \frac{W_{T_1} - W_P}{\dot{\Phi}_{2-3}} \cdot 100\% \tag{11}$$

where η_{t_R} represents the thermal efficiency of Rankine steam bottom cycle; W_{net} is the net output work of this bottom cycle.

(6) In this bottom cycle, the exhaust gas energy recovery efficiency is defined as effective output work divided by the engine exhaust gas energy. Firstly, the calculation formula for the flow rate of exhaust gas energy is given. As the specific heat of engine exhaust gas varies monotonically with exhaust gas temperature, the flow rate of exhaust gas energy can be calculated according to formula (12).

$$W_{ex} = q_{m\_e} \cdot \int_{T_0}^{T_{ex}} c_{P\_ex} \mathrm{d}T \tag{12}$$

where W_{ex} represents the flow rate of engine exhaust gas energy; c_{P_ex} is the specific heat at constant pressure of engine exhaust gas; T_{ex} is the temperature of engine exhaust gas at heat exchanger inlet; and T_0 is the temperature of ambient air.

Then, the definition of exhaust gas energy recovery efficiency is given as follows.

$$\eta_{re\_R} = \frac{W_{T_1} - W_P}{W_{ex}} \cdot 100\% \tag{13}$$

where η_{re_R} represents the exhaust gas energy recovery efficiency of Rankine steam bottom cycle.

Moreover, the steam expansion ratio is a very important parameter for selecting or designing the turbine since it determines the geometric dimension of turbine, and its calculation formula is given in formula (14).

$$\gamma = \frac{v_{out}}{v_{in}} \tag{14}$$

where, ν_{in} and ν_{out} are the specific volumes of steam at the inlet and outlet of turbine, respectively.

4.1.1.3. Boundary conditions and calculation of bottom cycle. Next, this bottom cycle was applied on a passenger car engine, and the bottom cycle performances as well as exhaust gas energy recovery potential were studied. The calculation and analysis of this bottom

cycle were based on engine common operating condition, and the related experimental data of engine exhaust gas was given in Table 6. As shown in the table, the engine exhaust gas temperature at heat exchanger inlet was 550 °C. The energy flow rate of engine exhaust gas was 14.8 kW at this operating condition, which could be calculated through energy conservation equation according to the mass flow rate and specific heat capacity of exhaust gas.

In the bottom cycle, the heat balance of heat exchanger makes much difference to the bottom cycle since it determines the heat transfer quantity from engine exhaust gas to bottom cycle working medium. According to the calculation formulas of heat exchanger and the design parameter *kA* given in Table 7, the performances of heat exchanger (boiler), including the heat exchanger NTU and heat exchanger efficiency ε, were calculated and also given in Table 7. Then, the heat transfer quantity can be acquired according to the boundary conditions. On this basis, the bottom cycle performances especially exhaust gas energy recovery potentials of Rankine steam bottom cycle were studied under lots of turbine inlet steam pressure and several kinds of turbine outlet steam pressure. All the state parameters were calculated by using the thermodynamic performance program of steam according to the initial conditions and process characteristics, and the bottom cycle performances as well as the exhaust gas energy recovery efficiency were calculated by programming on computer.

4.1.1.4. Calculation results and analysis. In the Rankine steam bottom cycle, steam pressure is one of the most important parameters. For this reason, the effects of steam pressure on bottom cycle performances were analyzed. Fig. 7(a) shows the relationship between cycle efficiency and steam pressure (including the turbine inlet steam pressure and outlet steam pressure). As can be seen from the figure, cycle efficiency increases with the turbine inlet steam pressure, while the increase rate gradually decreases. Meanwhile, a lower outlet steam pressure results in a higher cycle efficiency. At the inlet steam pressure of 3.0 MPa, the maximum cycle efficiencies are 17.6%, 19.1% and 20.5%, respectively, when turbine outlet steam pressure is 0.1 MPa, 0.07 MPa and 0.05 MPa. Fig. 7(b) illustrates the required volume expansion ratio of steam under different turbine inlet steam pressure and outlet steam pressure. As shown in the figure, the higher the turbine inlet steam pressure is, the larger the volume expansion ratio will be required. Although the larger steam volume expansion ratio results in higher cycle efficiency and exhaust gas energy recovery efficiency, the expansion ratio is limited by turbine structure size actually and engineering design, etc. With the installation space in automobile and geometrical structure of turbine considered, the expansion ratio of turbine is impossibly too large. Fig. 7(c) gives the output power of this Rankine steam bottom cycle, which is the recovered useful energy from engine exhaust gas. Actually, the output power depends on the cycle efficiency, and both of them have the same variation tendency. Finally, the energy recovery efficiency of this bottom cycle system was shown in Fig. 7(d). It rises gradually with the increasing of turbine inlet steam pressure but the growth rate becomes gradually smaller. Besides, bottom cycle has higher energy recovery efficiency under lower turbine outlet steam pressure. This is because turbine can recover more energy through steam fully expanding. However, the lower turbine outlet steam pressure requires higher steam volume expansion ratio, which is limited by the turbine geometrical structure. Furthermore, the reason why energy recovery efficiency is relatively low is that most of the exhaust gas thermal energy is used to evaporate the steam as latent heat, which can not be recovered by expansion. As a result, the maximum energy recovery efficiency of this bottom cycle is nearly 13.9% at the turbine outlet pressure of 0.1 MPa. However, over-heated Rankine steam cycle has great improvement and it would be discussed below.

Table 6
Engine exhaust gas boundary conditions.

Number	Items	Content
1	Speed (r/min)	2500
2	BMEP (MPa)	0.5
3	Exhaust gas mass flow rate (g/s)	23.81
4	Exhaust gas temperature at heat exchanger inlet (°C)	550
5	Exhaust gas specific heat (kJ/kg K)	1.185
6	Exhaust gas energy flow rate (kW)	14.8

Table 7
Designed parameters and calculated parameters of components in standard Rankine steam bottom cycle.

Number	Items	Content
1	Heat exchanger kA (W/K)	100
2	Pump efficiency, η_P	0.85
3	Turbine efficiency, η_T	0.75
4	Heat transfer effectiveness, η_h	0.95
5	Engine exhaust gas ($q_m c$) (W/K)	0.028
6	Heat exchanger, NTU	3.56
7	Heat exchanger efficiency, ε	0.97

4.1.2. Over-heated Rankine steam bottom cycle

4.1.2.1. Working principle and characteristics of over-heated Rankine steam cycle. In order to improve the exhaust gas energy recovery efficiency of Rankine steam bottom cycle, over-heated Rankine steam bottom cycle was proposed. The conceptual sketch and T–S diagram of this bottom cycle for exhaust gas energy recovery are shown in Fig. 8(a) and (b), respectively. Compared to standard Rankine steam bottom cycle, this bottom cycle has one more component: that is the superheater. In the superheater, the working medium steam is heated to over-heated state, and this process corresponds to the process 3–4 in the T–S diagram of Fig. 8(b).

Then, this bottom cycle concept was also applied on the passenger car engine, and the boundary conditions as well as the calculating method are the same as standard Rankine steam bottom cycle. As a result, the results of this two bottom cycle concepts are comparable. By the way, with the expansion ratio of turbine considered, the steam pressure at the turbine outlet is set to 0.1 MPa.

4.1.2.2. Calculation results and analysis. In this over-heated Rankine steam bottom cycle, the heat flux from engine exhaust gas to bottom cycle working medium is divided into two parts: one part of heat flux flows into the boiler and the other part flows into superheater. The allocation proportion depends on the mass flow rate of working medium. Fig. 9(a) shows the heat flux of engine exhaust gas flows into the boiler. As illustrated in this figure, heat transfer quantity of boiler increases with the mass flow rate of steam. In addition, the range of the mass flow rate of working medium is limited by the boundary conditions of bottom cycle. That is, the highest steam temperature should be lower than the engine exhaust gas temperature, and the lowest steam temperature should be higher than the saturated steam temperature. Actually, if the lowest steam temperature equals the saturated steam temperature, it means that the bottom cycle turns into standard Rankine steam bottom cycle. Under this circumstance, the heat flux of engine exhaust gas only flows into the boiler and the superheater does not work. Fig. 9(b) gives the relationship between the steam

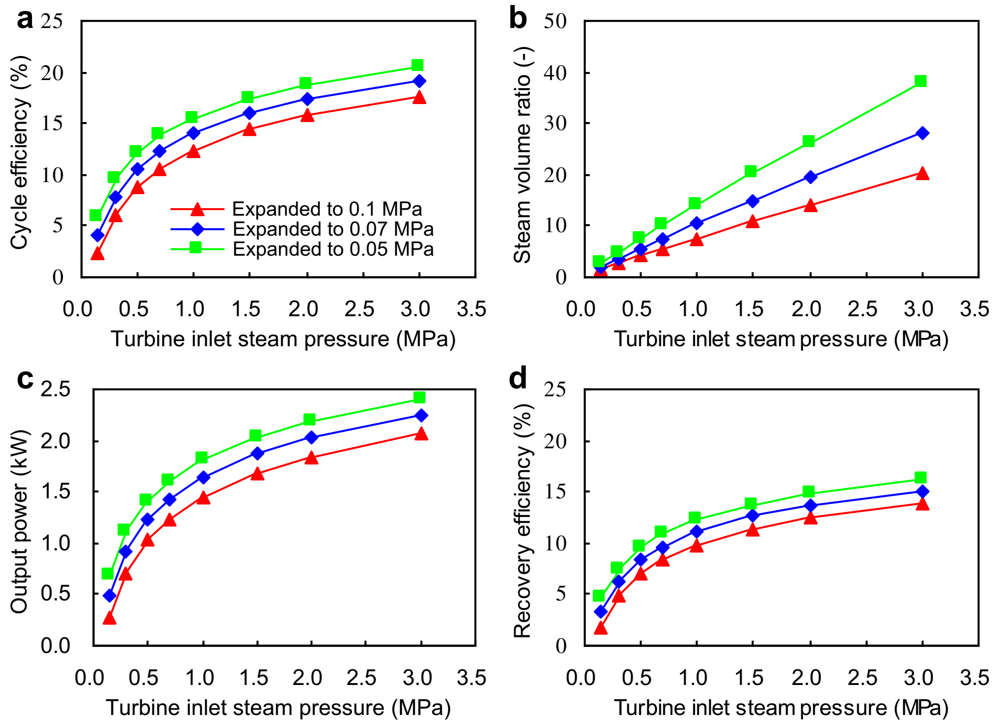

Fig. 7. Calculation results of standard Rankine bottom cycle.

temperature at the outlet of superheater and the mass flow rate of working medium. Also, it shows the available range of steam flow rate in this bottom cycle. As a result, the bottom cycle performances were analyzed under the whole available range of steam flow rate.

As shown in Fig. 9(c), the cycle efficiency of over-heated Rankine steam bottom cycle is obviously larger than that of standard Rankine steam bottom cycle. Compared to steam flow rate, steam pressure plays a more important role in the cycle efficiency. However, if the steam pressure reaches certain level, the effect of steam pressure on cycle efficiency will turn lower. When the steam pressure is 3.0 MPa, the maximum cycle efficiency can come up to 20.9%. Different cycle efficiency requires different steam expansion ratio. In order to obtain the cycle efficiency shown in Fig. 9(c), the required steam volume expansion ratio is given in Fig. 9(d). It illustrates that the required steam volume expansion ratio still increases with both the steam pressure and steam flow rate. With the increasing of steam flow rate, the steam temperature decreases (in other words, the degree of superheat is reduced), and the over-heated Rankine steam bottom cycle tends to standard Rankine steam bottom cycle. Consequently, on condition of the same steam pressure, the over-heated Rankine steam bottom cycle requires lower steam expansion ratio, which is beneficial to downsize the turbine. Then, the output power and energy recovery efficiency of over-heated Rankine steam bottom cycle are given in Fig. 9(e) and (f), respectively. It illustrates that: with the increasing of turbine inlet steam pressure, output power and energy recovery efficiency turn larger gradually. Compared to standard Rankine steam bottom cycle, over-heated Rankine steam bottom cycle has higher energy recovery efficiency due to higher steam temperature. However, both the maximum output power and energy recovery efficiency are limited by the fact that the higher steam temperature results in the lower working medium flow rate.

Through the above analysis, some conclusions can be drawn as follows.

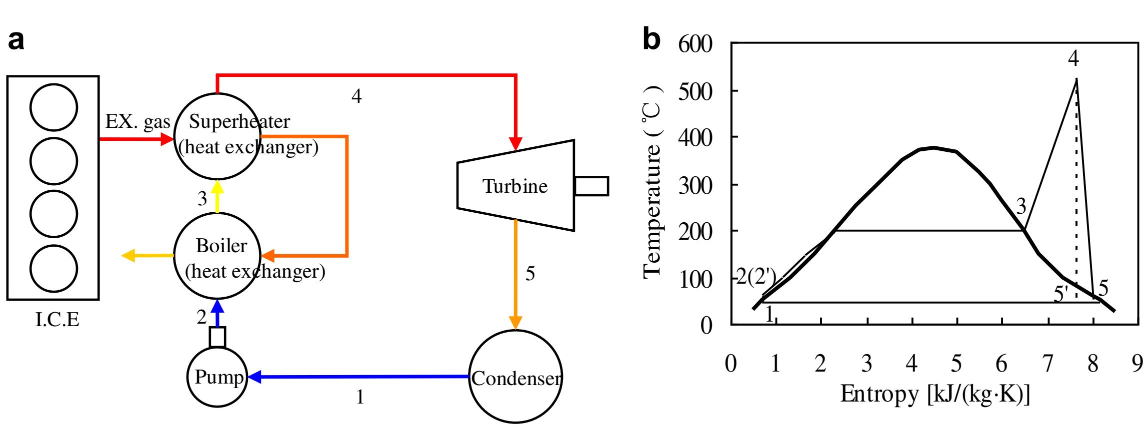

Fig. 8. Over-heated Rankine steam bottom cycle: working principle and T–S diagram.

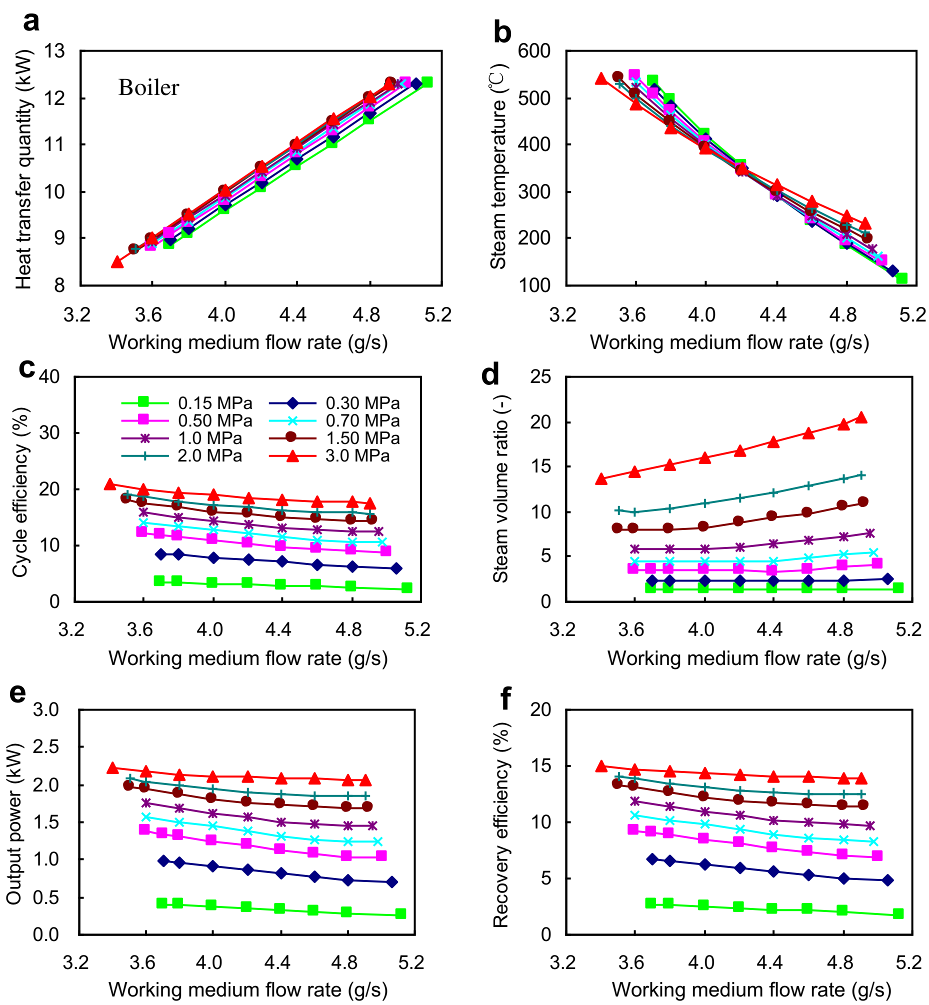

Fig. 9. Calculation results of over-heated Rankine steam bottom cycle.

In general, the cycle efficiency and energy recovery efficiency of over-heated Rankine bottom cycle are larger than those of standard Rankine steam cycle. The reason is that the over-heated steam has higher temperature than saturated steam. Moreover, over-heated Rankine steam cycle requires less steam expansion ratio. All these are very beneficial for bottom cycle system design. However, the cycle efficiency and energy recovery efficiency are still low. This is because latent heat of steam is too high and cannot be recovered by turbine expansion. As a result, organic medium with a low latent heat is expected to further improve the energy recovery efficiency and it will be studied later on.

4.2. *Indirect recovery bottom cycle: Brayton air cycle*

4.2.1. *Standard Brayton air cycle*

4.2.1.1. Working principle and characteristics of Brayton air cycle. - Brayton air cycle is another kind of typical thermodynamic cycle and it has been extensively utilized in various sceneries, such as aircraft and ship propulsion, gas and oil pumping and electricity generation [18–23]. In this paper, Brayton air cycle and its improved means were studied for engine exhaust gas energy recovery. First of all, standard Brayton air cycle was discussed. Fig. 10(a) is the conceptual sketch of standard Brayton air bottom cycle used for engine exhaust gas energy recovery. As shown in this figure, standard Brayton air bottom cycle consists of compressor, heat exchanger and turbine. In this bottom cycle, firstly, the working medium air is boosted to a certain pressure in the compressor; then, the working medium air is heated by the engine exhaust gas in the heat exchanger; next, the working medium air expands in the turbine and outputs effective work. All the thermodynamic processes were illustrated in the T–S diagram of Fig. 10(b). Being different from Rankine steam cycle, Brayton air cycle is only made up of three processes. As this bottom cycle takes air as working medium, the cycle system can be designed to an open cycle which does not require a condenser. Consequently, Brayton air bottom cycle is relatively simpler and more costeffective than Rankine steam cycle.

4.2.1.2. Thermodynamic processes analysis of Brayton air cycle. Combined with the T–S diagram of Brayton air cycle shown in Fig. 10(b), the thermodynamic processes of standard Brayton air cycle and main performance parameters were analyzed.

(1) In the compressor, working medium air is compressed to a certain pressure, and this process corresponds to process 1–2(2′) in the T–S diagram. Process 1–2′ represents the isentropic compression, while process 1–2 is the real compression process.

$$W_C = q_{m\_a}\cdot(h_{2'} - h_1)/\eta_C = q_{m\_a}\cdot(h_2 - h_1) \tag{15}$$

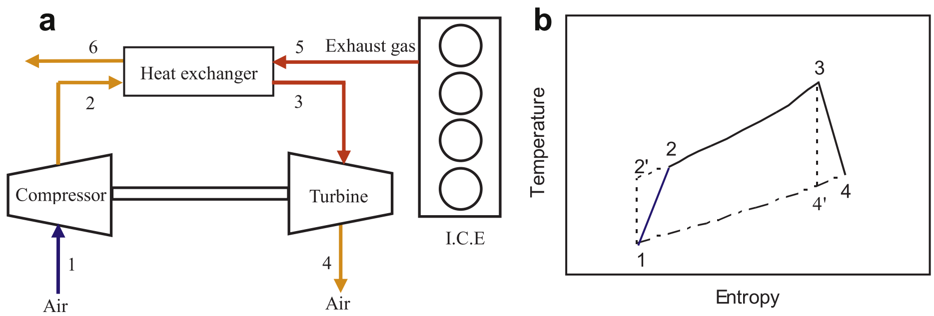

Fig. 10. Standard Brayton air cycle: working principle and T–S diagram.

where, W_C is the compression work consumed by compressor; q_{m_a} is the mass flow rate of air; η_C is the isentropic efficiency of compressor.

(2) In the heat exchanger, working medium air is heated by the engine exhaust gas, and it corresponds to the process 2–3 in the T–S diagram, which can be assumed to be an isobaric process. At the same time, the engine exhaust gas is cooled from state point 5 to state point 6. According to the energy conservation law, the following formulas are given.

$$\dot{\Phi}_{2-3} = q_{m\_a} \cdot (h_3 - h_2) \tag{16}$$

$$\dot{\Phi}_{5-6} = q_{m\_e} \cdot (h_5 - h_6) \tag{17}$$

$$\dot{\Phi}_{2-3} = \dot{\Phi}_{5-6} \cdot \eta_h \tag{18}$$

Since there is no phase change in the heat exchanger, the efficiency of counter flow heat exchanger can be calculated according to formula (7).

(3) In the turbine, working medium air expands and outputs effective work, and this process corresponds to process 3–4(4′). In the T–S diagram, process 3–4′ represents the ideal process (isentropic expansion), while process 3–4 represents the real process (irreversible expansion). And the output work of the turbine can be calculated according to formula (19).

$$W_{T_2} = q_{m\_a} \cdot (h_3 - h_{4'}) \cdot \eta_T = q_{m\_a} \cdot (h_3 - h_4) \tag{19}$$

where W_{T_2} is the output work of turbine in Brayton air bottom cycle.

(4) In the Brayton air bottom cycle, as the compressor is driven by the turbine, the net effective work of this bottom cycle equals the output work of turbine minus the compression work consumed by compressor. As a result, the calculation formula for the thermal efficiency of Brayton air bottom cycle is given by combining with the formulas (15), (16) and (19).

$$\eta_{t\_B} = \frac{W_{net}}{\dot{\Phi}_{2-3}} \cdot 100\% = \frac{W_{T_2} - W_C}{\dot{\Phi}_{2-3}} \cdot 100\% \tag{20}$$

where η_{t_B} is the thermal efficiency of Brayton air bottom cycle.

Finally, the exhaust gas energy recovery efficiency of Brayton air bottom cycle is defined in formula (21).

$$\eta_{re\_B} = \frac{W_{T_2} - W_C}{W_{ex}} \cdot 100\% \tag{21}$$

where η_{re_B} is the energy recovery efficiency of Brayton air bottom cycle.

Next, this bottom cycle concept was also applied on the same engine and at the same exhaust gas condition as Rankine steam bottom cycle. That is to say, the exhaust gas boundary conditions of Brayton air cycle and Rankin steam cycle are the same. By this means, the exhaust gas energy recovery potential of different kinds of bottom cycles are comparable. In the Brayton air cycle, the efficiency of compressor was assumed to be 0.78, and the initial parameters of heat exchanger (kA) were the same as Rankine steam cycle. The ambient air temperature was assumed to be 20 °C and the pressure was 1 bar. Similar to the Rankine steam cycle, the calculating processes of Brayton air cycle were realized by programming on computer.

4.2.1.3. Calculation results and analysis. Similar to Rankine steam bottom cycle, the main control parameters of Brayton air cycle are also the compression pressure and working medium flow rate. As the heat exchanger plays a very important role in this bottom cycle, its performance was analyzed at first. Unlike the Rankine steam cycle, heat exchanger efficiency in Brayton air cycle is more sensitive to the working medium flow rate. As shown in Fig. 11(a), it first descends and then ascends. The heat transfer quantity is determined by several parameters, including not only the heat exchanger efficiency, but also the mass flow rate and the compression temperature of working medium air. Although the heat exchanger efficiency fluctuates with the air flow rate, heat transfer quantity always increases, as shown in Fig. 11(b). Moreover, a higher compression pressure results in a lower heat transfer quantity due to the higher compression temperature.

The relationship between cycle efficiency and air mass flow rate is given in Fig. 11(c). As can be seen in this figure, cycle efficiency monotonically decreases with the air mass flow rate, while it first ascends and then descends with the compression pressure. This phenomenon is analyzed as follows. On the one hand, as the air mass flow rate rises, the air temperature decreases and then leads to expansion work of unit mass air descending. At the same time, the compression work consumed by compressor is a constant when the compression pressure of air is fixed. As mentioned above, net output work of bottom cycle equals the expansion work of turbine minus the compression work consumed by compressor. Thus, the net output work by unit mass air always decreases with the air mass flow rate. However, the total output power of bottom cycle is determined not only by the net output work of unit mass air, but also the mass flow rate of air. As a result, the two kinds of parameters lead to the result that the total output power first increases and then decreases with the air mass flow rate, as shown in

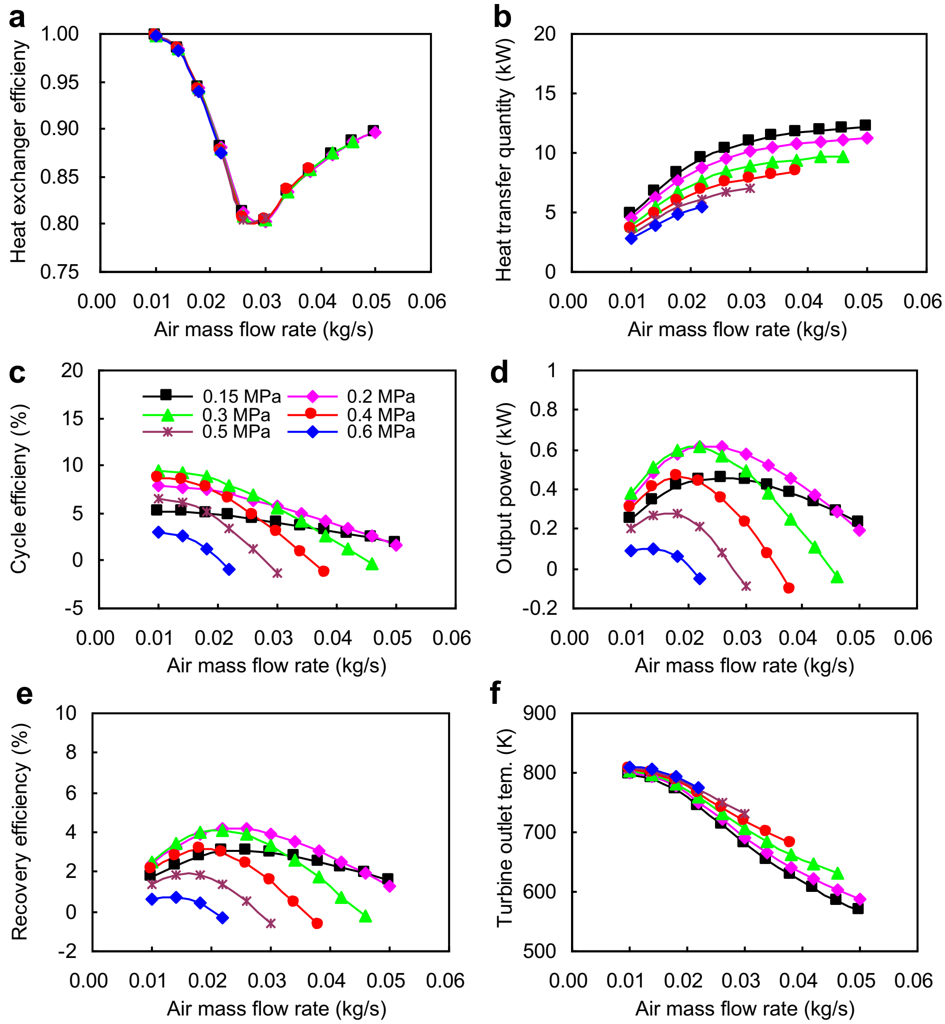

Fig. 11. Calculation results of standard Brayton air cycle.

Fig. 11(d), but the output power varies within a small range. On the other hand, the heat transfer quantity from engine exhaust gas to bottom cycle working medium always rises with the air flow rate. What's more, the growth rate of heat transfer quantity in heat exchanger is much larger than that of output work (if any), as shown in Fig. 11(b). Consequently, it results in the cycle efficiency monotonically decreases with the air mass flow rate. Meanwhile, when the air mass flow rate is fixed, with the increasing of air compression pressure, the heat transfer quantity will be reduced, and both the expansion work of turbine and the compression work consumed by compressor will increase. However, the growth rates of the two kinds of work differ from each other. Actually, the difference between the two kinds of work first increases and then decreases as the air pressure rises. Furthermore, when the compression pressure of working medium air comes up to certain level, the compression work consumed by compressor will become larger than the expansion work of turbine. As a result, there comes a negative value of bottom cycle output power when the compression pressure is higher than 0.6 MPa, as shown in Fig. 11(d).

Finally, the energy recovery efficiency of this standard Brayton air cycle is given in Fig. 11(e). It illustrates that energy recovery efficiency first increases and then decreases as the compression pressure and air mass flow rate rise, and the maximum energy recovery efficiency is nearly 4.2% when the compression pressure is around 0.2 MPa and air flow rate is 0.022 kg/s. Moreover, energy recovery efficiency of this standard Brayton air cycle is very low due to the poor cycle efficiency. The reason is that the effective output work of this bottom cycle depends on the difference between the expansion work of turbine and the work consumed by compressor.

Another reason for the cycle efficiency and energy recovery efficiency are very low is that the working medium air does not fully expand in the turbine, and the air temperature after expansion is still very high, as shown in Fig. 11(f). As a result, quite a number of thermal energy in the working medium air is wasted again. Based on this consideration, the standard Brayton air cycle was modified and the Brayton air cycle with regeneration was proposed for the purpose of recovering the waste heat energy of the turbine exhaust gas.

4.2.2. *Brayton air cycle with regeneration*

4.2.2.1. Working principle and characteristics of Brayton air cycle with regeneration. It is well known that regeneration could improve Brayton cycle efficiency and system performance. Next, Brayton air cycle with regeneration is also used for exhaust gas energy recovery. The conceptual sketch of Brayton air cycle with regeneration is shown in Fig. 12(a), and its T–S diagram is given in Fig. 12(b). Being different from the standard Brayton air cycle, this bottom cycle has a regenerator, and the working medium air can be

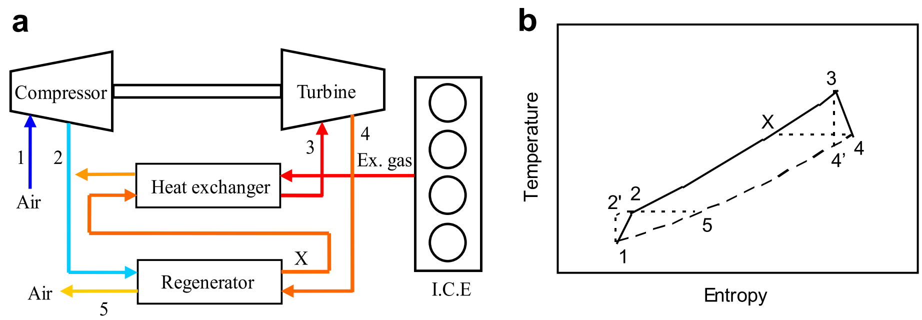

Fig. 12. Brayton air cycle with regeneration: working principle and T–S diagram.

preheated by turbine exhaust gas in the regenerator before it enters into the heat exchanger. By this means, part of the turbine exhaust gas energy can be also recovered.

Combined with Fig. 12(a) and (b), the processes of regenerated Brayton air cycle were described as follows. Process 1–2(2′): working medium air is compressed to a certain pressure in the compressor; in the T–S diagram, process 1–2′ represents the isentropic compression process while the process 1–2 is the real compression process. Process 2–X: working medium air is heated by the turbine exhaust gas in the regenerator. Process X–3: working medium air is further heated by the engine exhaust gas in the heat exchanger, and it can be regarded as an isobaric process. Process 3–4(4′): working medium air expands in the turbine and outputs effective work; in the T–S diagram, process 3–4′ represents the isentropic expansion process while process 3–4 is the real expansion process.

4.2.2.2. Calculation results and analysis. Fig. 13(a)–(f) shows the calculation results of Brayton air cycle with regeneration. Compared to the standard Brayton air cycle, Brayton air cycle with regeneration is more complex since there are two heat sources in the thermodynamic cycle. In addition, the heat transfer quantities of the regenerator and heat exchanger change with both the mass flow rate and compression pressure of working medium air, as shown in Fig. 13(a) and (b). In the regenerator, the heat transfer quantity first increases and then drops with the air flow rate, and it always decreases as the compression pressure increases. However, in the heat exchanger, the heat transfer quantity monotonously increases with the air flow rate and compression pressure. That is to say, regenerator plays a more important role in the range of low compression pressure. When the compression pressure is larger than 0.6 MPa, the "regeneration" will disappear. Under this circumstance, the bottom cycle is the same as standard Brayton air cycle. The efficiency of heat exchanger is also different from that of regenerator, as shown in Fig. 13(c). The regenerator efficiency always decreases with the air flow rate, while the variation trend of heat exchanger efficiency in regenerated Brayton air cycle is similar to that in the standard Brayton air cycle.

As the regenerator is used in the Brayton air cycle, the outlet temperature of turbine is changed. As it can be seen in Fig. 13(d), higher outlet temperature of turbine in the regenerated Brayton air cycle appears in the lower compression pressure, and it is different from that in the standard Brayton air cycle shown in Fig. 11(f). Fig. 13(d) shows the outlet temperature of regenerator, the variation trend of which is very similar to that of turbine outlet temperature. But outlet temperature of regenerator is lower than that of turbine since part of heat flux is recovered by regenerator. However, when the compression pressure reaches 0.6 MPa, they are nearly equal and it results in the phenomenon that heat transfer quantity in regenerator tends to zero, as shown in Fig. 13(a). Furthermore, when the compression pressure of working medium air is larger than 0.6 MPa, the outlet temperature of turbine will be lower than the compression temperature. Under these circumstances, the turbine exhaust gas will be heated by the intake air and the heat transfer quantity will come up negative value.

The cycle efficiency of Brayton air cycle with regeneration is much higher than that of standard Brayton air cycle especially under lower compression pressure. The highest value comes up to 21.6%, which appears at the pressure of 0.2 MPa. However, the improvement rate of bottom cycle output work is very limited, as shown in Figs. 11(d) and 13(g). As a result, it also leads to the improvement rate of energy recovery efficiency in regenerated Brayton air cycle is very little, as shown in Figs. 11(e) and 13(h). The most fundamental reason is that: although the cycle efficiency of Brayton air cycle with regeneration is improved significantly by the means of recovering turbine waste heat, the heat transfer quantity in heat exchanger is reduced at the same time. In other words, the cycle efficiency is improved, but it can't demonstrate the energy recovery efficiency and output power are also increased since the engine exhaust gas energy flows into bottom cycle is reduced!

4.3. Comparison of several typical bottom cycles

Finally, the cycle efficiency and energy recovery efficiency of standard Rankine steam cycle, over-heated Rankine steam cycle (turbine outlet steam pressures of the two kinds of Rankine steam cycles are 0.1 MPa), standard Brayton air cycle and Brayton air cycle with regeneration were compared, and the results are given in Fig. 14(a) and (b), respectively. From the viewpoint of bottom cycle efficiency, the standard Brayton air cycle has the lowest cycle efficiency because the large quantity of exhaust gas thermal energy recovered is wasted again through turbine exhaust gas since working medium air cannot expand fully in the turbine. Also, the compressive work consumed by compressor is too high. However, the Brayton air cycle with regeneration has the highest cycle efficiency because it has recovered the turbine exhaust gas energy. In addition, Brayton air cycle with regeneration requires the lowest compression pressure to achieve the highest cycle efficiency.

When from the viewpoint of energy recovery efficiency, the over-heated Rankine steam cycle has the highest energy recovery efficiency and the standard Rankine steam cycle has the second highest energy recovery efficiency. At the same time, the energy recovery efficiency of the two kinds of Brayton air cycles are very low, the main reasons can be listed as follows: on the one hand, the heat flux from engine exhaust gas to bottom cycle working medium in Brayton air cycle is smaller than that in Rankine steam cycle, because heat exchanger has higher efficiency and higher heat transfer quantity in Rankine steam cycle; on the other hand, the work consumed by compressor in Brayton air cycle is much larger than the work consumed by pump in Rankine steam cycle.

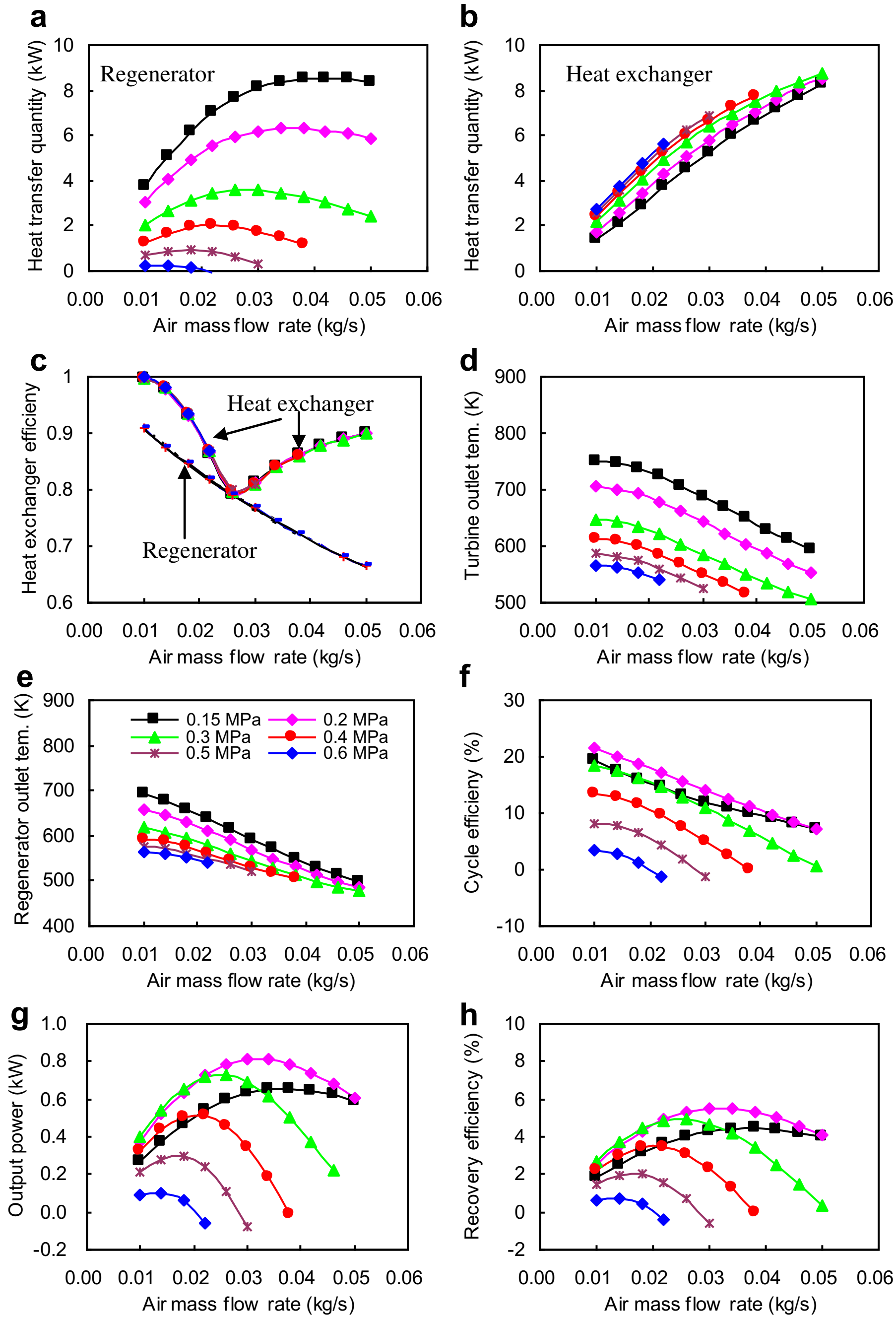

Fig. 13. Calculation results of Brayton air cycle with regeneration.

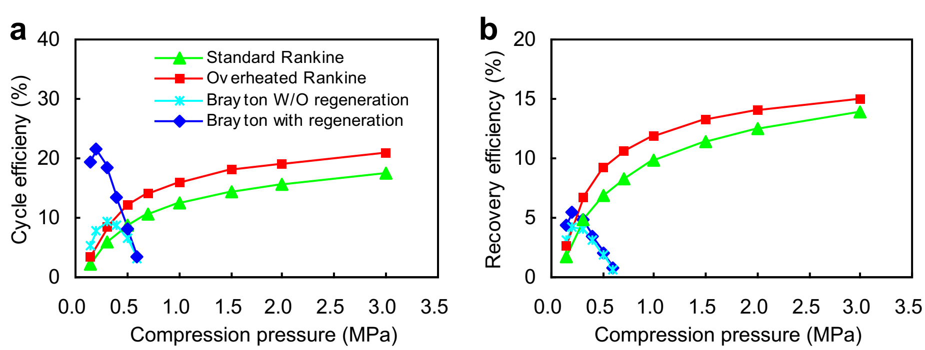

Fig. 14. Comparison of several typical bottom cycles.

Although Brayton air cycle with regeneration has the highest cycle efficiency, the heat transfer quantity of heat exchanger is much lower than that in Rankine steam cycle. As a result, its energy recovery efficiency is still very low.

5. Conclusions

In this paper, two major forms of engine exhaust gas energy recovery means, direct recovery and indirect recovery, were studied and various kinds of novel bottom cycles for engine exhaust gas energy recovery were proposed. Through analyzing and comparing the performance especially exhaust gas energy recovery potentials of various kinds of bottom cycles, some conclusions can be reached.

(1) Direct recovery of exhaust gas energy through secondary expansion of the exhaust gas demonstrates little, if any, positive potential for a gasoline engine. This exhaust gas energy recovery means only suits to diesel engine at full load with high boost pressure. Only when the engine is running at full load with higher boost pressure can the secondary expander concept start to show potential to improve engine thermal efficiency. The improvement range also differs with engine speeds. In general, there is less improvement at low speed but more at high speed. The concept is likely to work better for highly boosted engines in stationary applications, such as power generation plants.

(2) Indirect recovery through heat transfer decouples the engine pumping loop with the energy recovery bottom cycles and demonstrates a greater energy saving potential than direct recovery means because the main form of exhaust gas energy is thermal energy rather than pressure energy. In addition, indirect recovery has little influence on engine working cycle and can suit to both gasoline engine and diesel engine under a larger range of operating conditions.

(3) From the viewpoint of energy recovery efficiency, in all indirect recovery bottom cycles proposed in this paper, the energy recovery potentials reduce in the sequence of over-heated Rankine steam cycle, standard Rankine steam cycle, Brayton air cycle with regeneration and standard Brayton air cycle. The energy recovery efficiency of Rankine steam cycle is restricted by the fact that a large part of exhaust gas energy recovered is used to evaporate working medium and wasted as latent heat. Brayton air cycle is simpler than Rankine steam cycle since it is a kind of open cycle. However, its energy recovery efficiency is limited by the compression work consumed by compressor and the heat transfer quantity in heat exchanger.

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